2012

Impact of the Refrigerant Layout and Fin Cuts on the Performance of a Microchannel Condenser and a Gas Cooler

Santiago Martinez-Ballester
sanmarba@iie.upv.es

José Gonzálvez-Maciá

José M. Corberán

Impact of the Refrigerant Layout and Fin Cuts on the Performance of a Microchannel Condenser and a Gas Cooler

Santiago MARTÍNEZ-BALLESTER *, José GONZÁLVEZ-MACIÁ, José-M. CORBERÁN

Universitat Politècnica de València, Instituto de Ingeniería Energética,
Valencia, Spain
* Corresponding Author: sanmarba@iie.upv.es, Tel.: +34963879121, Fax: +34963879126

ABSTRACT

Microchannel heat exchangers (MCHX) play an important role in topics about charge reduction and transcritical cycles due to its high compactness and high mechanical strength. From a designer's point of view, given an air-side area and face area, there are many options to design the refrigerant circuitry and aspect ratio of a MCHX. The paper presents numerical studies about the influence on the MCHX's performance of these design parameters for a condenser working with propane and a gas cooler working with CO_2. A technique to improve the MCHX effectiveness is to cut the fins along the middle section between neighboring tubes. Following same methodology as was used in previous study, the paper analyzes the improvement by cutting fins for different condensers layouts. A key point of the paper is that the parameters analyzed can only be assessed by models which take into account heat conduction between tubes, otherwise their effects would be hidden. Results illustrate the presence of an optimum circuitry in order to maximize heat transfer of condensers working in similar conditions. Improvement on the condenser heat transfer by cutting fins depends on the layout which turned to as much as 4%. Impact of heat conduction between tubes was higher and clearer in case of gas cooler.

1. INTRODUCTION

Currently, an increasing interest on microchannel heat exchangers (MCHXs) is arising in refrigeration and air conditioning applications due to their high compactness and high effectiveness. The high effectiveness is consequence of large heat transfer coefficients as result of using small hydraulic diameters. Given an air side heat transfer area, high compactness means reduced volumes what will result in light heat exchangers and low refrigerant charges.

Getting low refrigerant charges plays an important role on the use of natural refrigerants which are flammable like propane. Natural refrigerants are considered as more environmentally friendly than others refrigerants commonly used, with a similar or even better performance. The main drawback of working with some natural refrigerants is that they are dangerous in large quantities: ammonia is toxic and propane is highly flammable, in fact IEC 60335-1 restricts the amount of a hydrocarbon that can be used in a system to 150 g. To this end, a suitable heat exchanger design is a serpentine MCHX. This kind of heat exchangers minimizes the refrigerant charge because it has no headers, thus saving all this volume and the corresponding refrigerant charge. In the case of transcritical CO_2 systems, microchannels have an additional merit related to their high mechanical strength.

Along the design process of a MCHX, geometric data of tubes and fins are usually imposed by manufacturer. Fin pitch, heat transfer and face areas of a MCHX are usually obtained attending to performance requirements. However, multiple choices exist for the number of refrigerant passes, refrigerant connections and aspect ratio (L/H) of the MCHX. In fact, some simulation software like EVAP-COND (2003) has the capability to optimize the heat transfer by varying the circuitry of a finned tube heat exchanger. Shao et al. (2009) studied the effect of the number of refrigerant passes for a serpentine MCHX working as a condenser, with the same face area and heat transfer area.

The authors obtained up to 30% differences on heat load only by changing the number of refrigerant passes. Since the circuitry has an important influence on the heat exchanger performance, the usefulness of a simulation software for this purpose is clearly justified because the optimization via experimentation takes too long, it is difficult and expensive.

On the other hand, depending on the model's assumptions some parameter can be studied or not. Most of models for air-to-refrigerant heat exchangers apply the fin theory by using the analytical solution for adiabatic-fin-tip assumption. In this way, these models do not account for heat conduction between tubes. Performance results of these approaches are not affected by the parameters studied in the present paper. For instance, the impact of the aspect ratio (L/H) on the heat transfer of a heat exchanger would be null if it is evaluated with such models. This effect can only be assessed if the model adequately accounts for the heat conduction between tubes.

The authors presented a detailed model referred to as Fin2D model (Martínez-Ballester et al., 2011a) in order to analyze the influence of these classical assumptions in a microchannel gas cooler. The air-side the heat transfer was modeled without applying any of these assumptions but it required a detailed discretization and a large computational cost. The conclusions of that work allowed continuing working in a model which could retain only the most important effects but with an interesting ratio between accuracy and computational cost. The result was the Fin1Dx3 model (Martínez-Ballester et al., 2011b). The main difference of this approach is the discretization applied to the air and fin elements. This approach reduces one order of magnitude the simulation time with regard the Fin2D model retaining same accuracy. Fin1Dx3 model is the model employed to carry out the numerical studies presented in this paper.

Cutting the fins between tubes for air-to-refrigerant heat exchangers is an improvement studied in literature. Cutting the fins avoids the heat conduction between tubes along the fins, which degrades the heat exchanger effectiveness. Heat conduction between tubes along fins appears when a temperature difference between tubes exists. Several experimental studies indicated that the heat exchanger performance can be significantly degraded by the tube-to-tube heat transfer via connecting fins. Domanski et al. (2007) measured capacity reductions as much as 23 % in a finned-tube evaporator when different exit superheats were imposed on individual refrigerant circuits. However, not so large improvements have been achieved for MCHXs, namely: Asinari et al. (2004) concluded that the impact of this heat conduction can be assumed as negligible in a wide range of applications; Park and Hrnjak (2007) reported measurements of capacity improvements of up to 3.9% by cutting the fins in a CO_2 serpentine microchannel gas cooler.

Numerical studies of the parameters analyzed in this paper, for a MCHX, are hardly available in the literature. The goal of the selected case studies is contributing to a better understanding of the influence of some design parameters on performance of MCHXs.

2. MODEL DESCRIPTION

The model used for the present work applies the Fin1Dx3 approach (Martínez-Ballester et al., 2011b). The model can simulate any refrigerant circuitry arrangement: any number of refrigerant inlets and outlets, and any connection between different tube outlets/inlets at any location. Fig. 1(a) shows a MCHX that can be simulated with this model.

First, the heat exchanger is chopped into segments along the X direction (refrigerant flow), resulting N_s segments per tube. The discretization for each segment is the same and it is shown in Fig. 1(b). Each segment consists of: a refrigerant stream that is split into N_r channels in the Z direction; a flat tube that is discretized into N_{tw} cells in the Z direction; and both air flow and fins, which are discretized in two dimensions: three cells in the Y direction and N_a cells in the Z direction. In this way, it allows to capture the most significant air temperature values: the air temperature close to each tube and the bulk air temperature. Since the discretization for the air and fin wall is the same, $N_{fw}= N_a$.

The Fin1Dx3 approach applies a novel methodology to model the air side heat transfer. Fin1Dx3 adopts a piecewise function for the fin wall temperature which applies the fin theory for each fin-to-air connection by using the analytical solution of the fin theory for the case of given different fin roots temperature. Thus heat conduction between tubes along the fin is taken into account. Finally, the specific discretization in the air side captures the

unmixed air (along *Y* direction) influence in the air-to-tube heat transfer evaluation. For more details regarding governing equations and its discretization, reader is referred to Martínez-Ballester et al. (2011b).
The main advantage of the proposed approach is that the heat conduction between tubes is taken into account implicitly without requiring application of more or less artificial correction terms to account for this phenomenon, as other current models do.

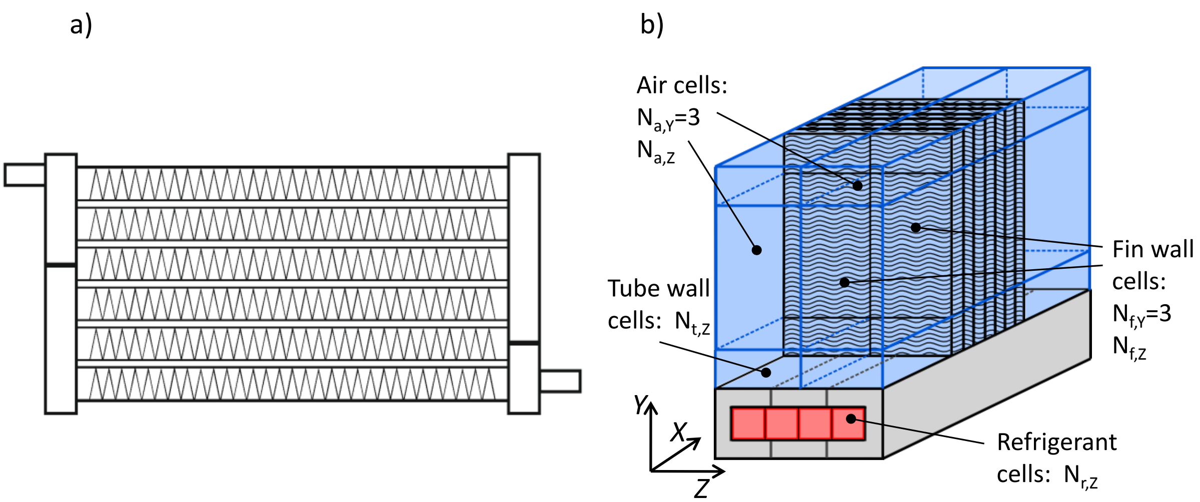

Figure 1: (a) Microchannel heat exchanger of three refrigerant passes. **(b)** Schematic of a heat exchanger segment.

3. SIMULATION STUDIES

3.1 Case Study

The MCHX condenser corresponds to a condenser working with propane as working fluid. The baseline condenser is a serpentine condenser which consists of one circuit and 12 passes. Rest of geometry for tubes and fins is shown in Table 1.

Table 1: Geometric characteristics of the microchannel condenser

Face area (cm^2)	350	Refrigerant side area (cm^2)	2402
Tube length (mm)	275	Tubes number of tubes	12
Fin type	Louvered	Core depth (mm)	19
Number of ports	19	Fin pitch (mm)	2.54,1.81,1.2
Wall thickness (mm)	0.43	Port hydraulic diameter (mm)	0.83
Fin thickness (mm)	0.1	Fin height (mm)	8.1

Table 2: Operating conditions for condenser.

Fluid	Inlet Pressure (kPa)	Inlet temperature (°C)	Mass flowrate (g/s)
Propane	1400	70	1.8
Air	100	27	92.8

In order to study the impact on performance of the refrigerant layout, the number of refrigerant passes is a parameter to be studied. Number of refrigerant passes is changed from the baseline condenser (serpentine) up to a complete parallel flow arrangement (12 tubes with just one refrigerant pass). For all the scenarios the refrigerant, air-side and face areas were the same as well as rest of the geometry.

Inlet conditions for both fluids in the condenser are going to be identical for all simulation studies, which are shown in Table 2. Regarding the air, different scenarios were analyzed by varying the fin pitch between 10 and 21 fins per inch.

In order to get a better understanding, results for a gas cooler working with CO_2 under trasncritical pressure are also shown. In a gas cooler, the refrigerant does not undergoes a phase change, so a priori results are easier to understand than for a condenser. In a condenser due to the phase change pressure losses have also an impact on the heat transfer. The gas cooler results correspond to those presented by Martínez-Ballester et al (2012). Despite geometry and operating conditions are not the same, it is showed just to analyze differences in trends with regard to a condenser. Full description of geometry and operating conditions for the gas cooler studies are described in Martínez-Ballester et al (2012). Basically, the gas cooler geometry was almost identical to the condenser analyzed.

3.2 Number of Refrigerant Passes

The number of refrigerant passes is varied from 1 pass up to the maximum possible number; 12 passes which corresponds to a serpentine configuration. Fig. 2 depicts two samples of the cases studied. The performance differences will be only due to the number of passes since refrigerant area, air-side area, face area and rest of the geometry do not change.

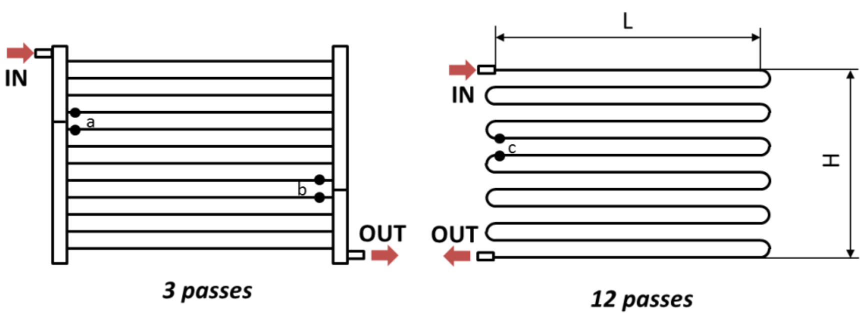

Figure 2: Schematics of two MCHX arrangements studied: 3 and 12 refrigerant passes.

Fig. 3(a) shows the results of this study for the condenser and different values of fin pitch. First fact that is noticeable is that, given a number of passes, heat transfer increases asymptotically with the fin pitch. It is asymptotic because the number of thermal units (NTU) of the condenser is already quite large for 14 fpi. An increase of the fin pitch and therefore in the NTU only reports a slight increase in capacity.

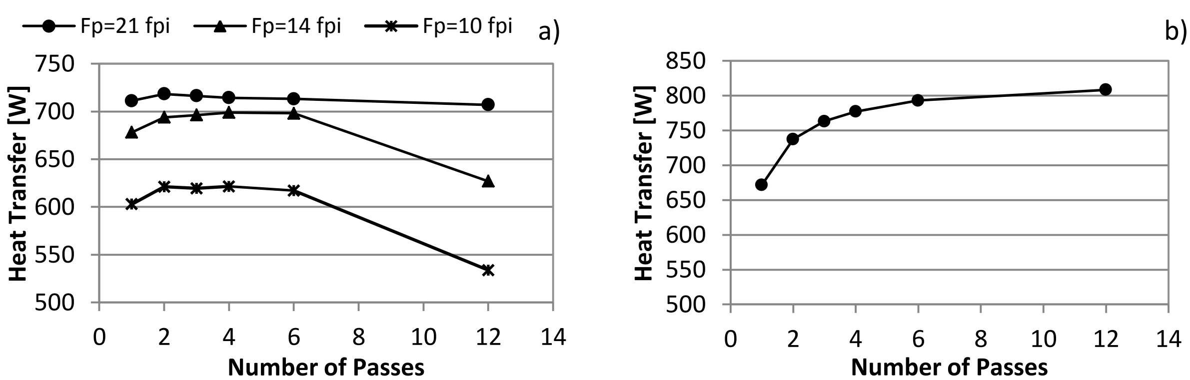

Figure 3: Heat transfer with continuous fin when number of refrigerant passes is changed for: **(a)** condenser with different fin pitch, **(b)** gas cooler.

When the number of passes is increased, the total refrigerant cross area is reduced so that the refrigerant velocity rises to keep constant the mass flow rate, and it improves the heat transfer coefficient. Thus, results for 10 fpi show that heat transfers increases from parallel flow to 2 pass arrangement. However increase of passes up to 4 does not report an increase on heat transfer. If number of passes is even more increased until the serpentine configuration, heat transfer decreases.

If a gas cooler is analyzed (Fig. 3(b)), the heat transfer is always raised with an asymptotic trend, by increasing the number of passes. The reason to these differences in results for condenser and gas cooler is that a gas cooler does not undergo a phase change. In a condenser, the pressure drop leads a temperature drop during the phase change, therefore the temperature difference between air and refrigerant would decrease and the heat transfer would be reduced. In this way, in condensers/evaporators there is an optimum on the heat transfer when the number of refrigerant passes is studied according to the opposite influence on the heat transfer and pressures drop. This conclusion was also exposed by Shao et al. (2009) in their studies for a serpentine microchannel condenser, where they studied the influence of the number of passes on heat transfer.

Heat transfer improvement by varying the number of refrigerant passes is less than 5% for 10 fpi. It could be thought that change in heat transfer would be larger if refrigerant thermal resistance had more impact. Usually, in air-to-refrigerant heat exchanger the dominant thermal resistance is on the air. Thus any change in refrigerant side has less impact on global capacity. To this end, fin pitch was increased in Fig. 3(a) in order to reduce air side thermal resistance. However maximum variation was still lower. A possible reason for that was pointed out above, which consists in the large NTU that baseline condenser has.

3.3 Influence of Cutting the Fins

A technique to improve the effectiveness in air-to-refrigerant heat exchangers is by cutting the fins. The heat conduction between tubes, due to temperature differences from bottom to top fin roots, degrades the heat exchanger effectiveness. By cutting the fins, this heat conduction is avoided. This technique is indicated for heat exchangers which have large temperature differences between tubes. For example, in a condenser there are tubes with superheated vapor flowing inside which are connected through fins to other tube with saturated vapor inside. Under these conditions large temperature differences can be expected. An extreme case corresponds to a gas cooler arrangement, in which the refrigerant undergoes a temperature variation along all the gas cooler length, since there is no phase change. Thus the temperature difference between two neighboring tubes can be as large as 50 K.

There only exist few models that take into account the heat conduction between tubes. The rest of models always overpredict the heat transfer for the same conditions since they do not account for the effectiveness degradation caused by the heat conduction.

In the present study the fin cuts studied are disposed along the middle section between two neighbour tubes for all the fins of the heat exchanger. The Fin1Dx3 model is developed for a continuous fin, but can be slightly modified to incorporate a cut in the section at half the fin height. To the authors' knowledge there are no studies for MCHXs about the influence of the refrigerant circuitry on the impact of fin cuts in the predicted results. To this end, the impact of cutting the fins has been evaluated for the same refrigerant passes studied in previous subsection.

For the condenser the results are shown in Fig. 4(a), where it has been plotted the heat transfer improvement by cutting fins with respect to the solution without fin cuts. The heat improvement for one pass is zero because in a one pass arrangement all tubes have the same temperature evolution resulting in a null temperature difference between tubes at the same X coordinate. In such a case the adiabatic-fin-tip assumption is fundamentally correct.

Regardless the fin pitch, the improvement on heat transfer by cutting the fins is larger as the number of refrigerant passes rises. There are two main reasons; while the heat transfer is enhanced by increasing the number of refrigerant passes (from 1 to 2 passes in Fig. 3(a)) the subcooling increases, which changes from no subcooling to 5 K. If subcooling is larger, heat conduction between tubes takes place due to the temperature difference between tubes and it degrades the effectiveness. However, when the capacity decreases by increasing the number of passes (from 2 up to the serpentine case in Fig. 3(a)) the pressure drop means a temperature drop and heat conduction between tubes takes place again. Hence, there is an improvement by using the fin cut arrangement which is not affected by these temperature differences.

In Fig. 4(a), the case of 21 fpi has a trend quite different from rest of cases. It looks to be rather related with the high value of NTU of such a configuration than with the effect of heat conduction between tubes. Anyhow is hard to explain such a trend since in the condenser case several phenomena is taking place at same time.

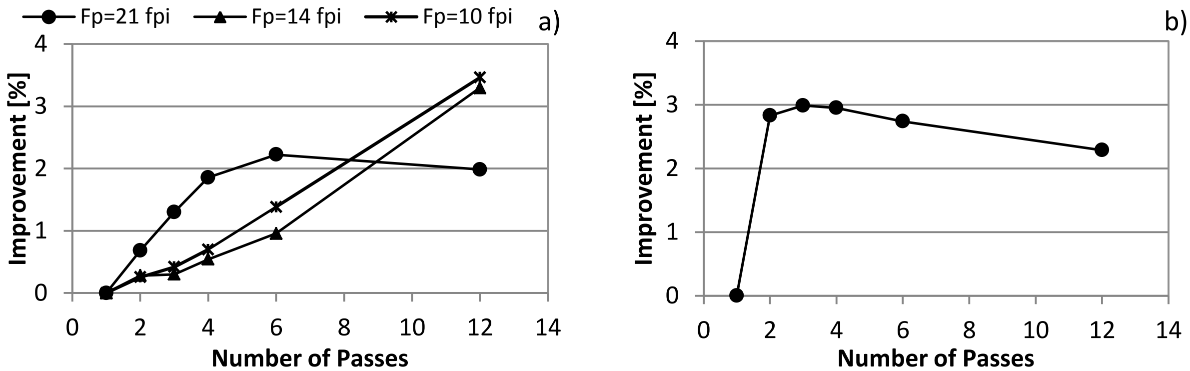

Figure 4: Improvement of heat transfer by cutting fins with respect to continuous fin for different number of refrigerant passes, **(a)** condenser with different fin pitch, **(b)** gas cooler.

If we analyze the case of gas cooler, Fig. 4(b), when the number of passes is different from one, there is always an improvement on the heat transfer by cutting the fins, and in this case there is a maximum value for 3 passes. When the number of passes is two, the fin roots which connect two tubes of different passes (central tubes of the heat exchanger) have a large temperature difference that produces a heat conduction flux. As the number of passes is increased the temperature difference between tubes decreases, but the number of fins with such a temperature difference rises. Fig. 2 illustrates this explanation, where the heat exchanger with 3 passes has two zones with large temperature difference, regions "a" and "b". The serpentine heat exchanger has a similar temperature difference between all the tubes, which can be represented by the temperature difference at zone "c". Heat exchanger with 3 passes will have only two zones with temperature difference, but the temperature difference between bottom and top of zones "a" or "b" is much higher than corresponding value for region "c" of the serpentine case, though serpentine MCHX has 11 regions with a similar temperature difference to the "c" zone. These opposite effects could be one of the reasons to explain the presence of a maximum in the heat improvement depicted in Fig. 4(b).

For the gas cooler case, the maximum improvement is as much as 3%. Similar values were reported by Park and Hrnjak (2007) who measured capacity improvements of up to 3.9% for a serpentine gas cooler. The comparison with the condenser case shows that improvement by cutting the fin is higher for the gas cooler case than for the condenser, excepting the serpentine case. This conclusion could be expected due to the large temperature difference between tubes that a gas cooler undergoes since there is no phase change.

3.4 Influence of aspect ratio for a serpentine gas cooler

A serpentine MCHX corresponds to a MCHX with a single tube which is bended in order to get a specific number of refrigerant passes. It has the particularity of not having headers therefore it is highly recommended for saving refrigerant charge thanks to its reduced internal volume.

A restriction for these studies is that air-side and face area are constants while aspect ratio (L/H) changes. By observation of the serpentine MCHX design, it is deducible that the air-side heat transfer area is proportional to the product: $N\ L$. Therefore, to study the isolated effect of the aspect ratio on the performance, $N\ L$ will have to remain unchanged for all the cases studied.

Fig. 5(a) and (b) show the results for the predicted heat transfer as function of the aspect ratio for condenser and gas cooler respectively. For condenser, only the case of 14 fpi was simulated. The figures show the results for the two analyzed cases: the fin is cut and a continuous fin. Regardless the MCHX is a condenser or a gas cooler, the figures show that aspect ratio has no effect on heat transfer when the fin is cut, thus models that apply adiabatic-fin-tip assumption will not be able to study this influence since results are always the same.

For the case of continuous fin, results of condenser and gas cooler are quite different. Fig. 5(a) shows the presence of an aspect ratio which minimizes the heat transfer. However, heat transfer difference for the different aspect ratios is very small. Reason could be that changes in the temperature field by modifying the aspect ratio are very small, so that the improvement is negligible.

For gas cooler scenario, Fig. 5(b) shows that heat transfer has a strong dependence on the gas cooler aspect ratio. It shows that is preferable to use a short heat exchanger length (large number of passes) instead of a large length (few passes). For the case analyzed, the highest value of aspect ratio corresponds to N=2 while lowest value correspond to N=16. An interesting observation is that the asymptote looks to be the capacity for the fin cut case. This fact means that the aspect ratio which maximizes the heat transfer corresponds to the value that minimizes the heat conduction between tubes.

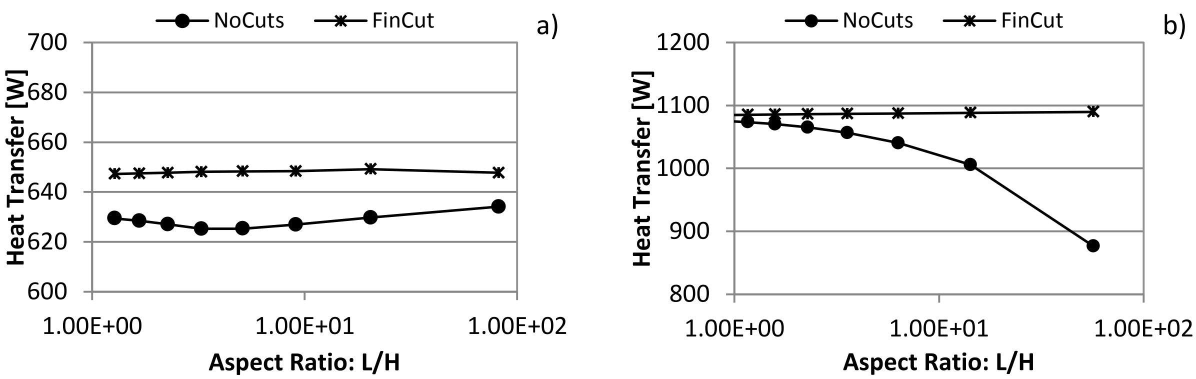

Figure 5: Heat transfer of the MCHX when the aspect ratio is varied for two modeling scenarios: continuous fin and fin with cuts in **(a)** condenser, **(b)** gas cooler.

Reason for the huge difference between both scenarios is may be due to the large difference in the temperature drop that takes place in the gas cooler scenario. Notice that for the conditions studied, the only influence of the aspect ratio is on the heat conduction between tubes. Pressure drop did not affect to the results because it resulted to be the same for all the cases since the refrigerant cross area and the total length are always the same.

4. CONCLUSIONS

From the designer's point of view, some parameters of MCHX such as number of refrigerant passes, aspect ratio and effect of fin cuts, are hard and expensive to determine by experimentation. On the other hand, success of simulation tools to this end depends on the model's assumptions, i.e. some parameters produce effects that cannot be assessed with a model that do not take into account the phenomena involved.

The proposed numerical studies were carried out by using a model for MCHX which uses a novel approach (Fin1Dx3) that takes into account: heat conduction between tubes, fin cut or continuous fin and effects of non-mixed air along Y direction.

For the studies carried out for both condenser and gas cooler, the main conclusions of the simulations studies were:

- For a gas cooler where no phase change occurs, heat transfer is always increased by increasing the number of refrigerant passes regardless the increase of pressure drop. However, in case of a condenser, pressure drop results in a saturation temperature drop, therefore there is an optimum circuitry which maximizes the heat transfer.
- The fin cuts always increase the heat transfer. In the gas cooler analyzed, there exists an optimum number of passes that maximizes the capacity improvement with regard to the continuous fin. For the condenser case, the trend depends on the air thermal resistance. Excepting for extreme cases, the gas cooler reports larger improvements than condenser by cutting the fins. Anyway, the improvement is as much as 3%.

- When the aspect ratio is analyzed, given the face and heat transfer areas, the best aspect ratio corresponds to a configuration that reduces the heat conduction between tubes. In case of gas cooler, the dependence is quite clear and the highest capacity corresponds to a design with reduced length (L) and large height (H). For the condenser, since the pressure drop is rather the same a change in the aspect ratio produces only slight changes in temperature field and its effect is negligible.
- Overall, the impact of the parameters analyzed depends on the effect of heat conduction between tubes for each scenario. The effect of heat conduction between tubes depends on the temperature drop along the fluid path and temperature distribution in the heat exchanger. Thus, the gas cooler scenario shows the largest impact by modifying the analyzed parameters.
- Dependence of gas cooler performance on the analyzed parameters is always asymptotic, while for condenser it has a minimum or maximum due to effects of pressure drop on heat transfer.

REFERENCES

Asinari, P., Cecchinato, L., Fornasieri, E., 2004, Effects of Thermal Conduction in Microchannel Gas Coolers for Carbon Dioxide, *Int. J. Refrigeration*, 27(6): 577-586.

Domanski, P.A., Choi, J.M., Payne, W.V., 2007, Longitudinal Heat Conduction in Finned-Tube Evaporator. *22nd IIR International Congress of Refrigeration, Beijing, China*.

EVAP-COND, 2003, Simulation Models for Finned Tube Heat Exchangers, http://www.nist.gov/el/building_environment/evapcond_software.cfm.

IEC 60335-1, 2010, Household and similar electrical appliances - Safety - Part 1: General requirements.

Martinez-Ballester, S., Corberan, J.M., Gonzálvez-Macia, J., Domanski, P.A, 2011a, Impact of Classical Assumptions in Modelling a Microchannel Gas Cooler, *Int. J. Refrigeration*, 34 (8): 1898-1910.

Martinez-Ballester, S., Corberan, J.M., Gonzálvez-Macia, J., 2011b, Numerical Model for a Microchannel Gas Cooler with Any Refrigerant Circuitry, *23rd IIR International Congress of Refrigeration, Prague, Czech Republic*.

Martinez-Ballester, S., Corberan, J.M., Gonzálvez-Macia, J., 2012, Impact of Fin Cuts and Refrigerant Layout on the Performance of a Microchannel Gas Cooler Working with Transcritical CO_2, *10th IIR Gustav Lorentzen Conference on Natural Refrigerants, Delft, The Netherlands*.

Park, C.Y., Hrnjak, P., 2007, Effect of Heat Conduction through the Fins of a Microchannel Serpentine Gas Cooler of Transcritical CO2 System, *Int. J. Refrigeration*, 30 (3): 389-397.

Shao, L.L., Yang, L., Zhang, C.L., Gu, B., 2009, Numerical Modeling of Serpentine Microchannel Condensers, *Int. J. Refrigeration*, 32 (6): 1162-1172.

ACKNOWLEDGEMENT

First author's work on this project was partially supported by Ministry for Education of Spain, under the training for university professors program (FPU). Financial support from Ministry for Education of Spain, project numbers: DPI2008-06707-C02-01 and DPI2011-26771-C02-01, is also gratefully acknowledged.